Experimental investigation into the influence of cross-winds on the performance of dry-cooling towers

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Abstract

Relatively little information is available in the literature quantifying the effect of cross-winds on the heat rejection rate of natural draft dry-cooling towers. In this study a series of tests are performed on a scale model of a circular natural draft dry-cooling tower with the heat exchangers arranged in a horizontal pattern (not A-frames) over the entire inlet cross-section of the tower. The effect of the wind on such towers is found to be complex and is influenced by a number of parameters including the wind velocity, the shape of the approaching wind profile, the heat exchanger and support pressure loss coefficients, the shape of the tower shell, the inlet diameter to inlet height ratio of the tower, the tower height and the heat rejection rate of the tower. Further reductions in the heat rejection rate of the tower are caused by a non-uniform air temperature distribution inside the tower and flow distortions through the heat exchangers.

Nomenclature

A	Area, m²
b	Exponent
C	Coefficient
C_D	Drag coefficient
c_p	Specific heat at constant pressure, J/kgK
d	Diameter m
H	Height, m
K	Loss coefficient
k	Roughness height, m
L	Length, m
Ny	Characteristic heat transfer parameter, m^{-1}
n	Number
p	Pressure, N/m²
$\triangle p$	Pressure differential, N/m²
α_Q	Heat transfer rate, W
Re	Reynolds number
Ry	Characteristic flow parameter, m^{-1}
r	Radius, m
T	Temperature, °C or K
U	Overall heat transfer coefficient, W/m²K
v	Velocity, m/s
z	Elevation, m
α_Q	Heat transfer correction factor
$\in$	Effectiveness
θ	Angle or circumferential position, °
μ	Dynamic viscosity, kg/ms
ρ	Density, kg/m³

Subscripts

a	Air
c	Cone
ct	Cooling tower
e	Effective
fr	Frontal
he	Heat exchanger
i	Inlet
l	Local
o	Outlet
p	Pressure
ref	Reference
t	Throat
ts	Tower supports
w	Wind or water
θ	Circumferential position
∞	Free stream

Abbreviations

arr	Arrangement
ex	Exchanger
max	Maximum
min	Minimum

Introduction

It is well known that cross-winds reduce the heat rejection rate of wet as well as dry natural draft cooling towers. Measurements performed on full scale natural draft dry-cooling towers indicate a rise in water temperature as the wind speed increases for a given heat rejection rate [1, 2, 3, 4, 5]. These results suggest that cross-winds affect certain dry-cooling towers more than others. Generally it would appear that towers where the heat exchangers are arranged horizontally in the inlet cross-section are less affected by cross-winds than those where the heat exchangers are arranged vertically around the circumference of the tower. The following factors may also contribute to the wide scatter in the data:

1. The extent of disturbances caused by wind is not only a function of the wind velocity, but also of the air velocity through the heat exchangers, which is directly related to the tower performance.
2. The unstable character of the wind, both in speed and direction may cause scatter in the data [6].
3. Since significant variations in the wind velocity may

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be found in the surface boundary layer, it is important to know the wind velocity distribution, as well as the position and height above ground level where the wind velocity and the inlet air temperature are measured during any test.

4. Witte [7] mentioned that the influence of winds on a cooling tower can only be tested reliably in the absence of other atmospheric disturbances. Accordingly it is suspected that the measurements presented in at least some publications may be affected by other disturbances like temperature inversions [4].

Due both to all the problems which arise during full scale tests, and to practical and economical considerations, some investigators [8, 9, 10, 11, 12, 13] preferred model tests for studying the wind effect on cooling towers. These tests were either based on Froude's similarity or on isothermal tests approximating Reynolds similarity, because it is impossible to satisfy all the relevant dimensionless groups with a single model test. Unfortunately, in most of the abovementioned papers, only a single aspect of the problem was addressed or a model of a specific dry-cooling tower was used to perform the tests. Furthermore very few of the authors [7, 13, 14] used their results to predict the effect of a cross-wind on the heat rejection rate of a full scale tower.

Witte [7] suggests that the wind effect on a cooling tower can be predicted by using the pressure distribution around the outside of the tower shell. The results obtained by this method agree well with full scale measurements as published by Markòczy [4]. Witte, however, totally ignored the wind effect at the outlet of the tower, while others [10, 11, 12, 13, 15, 16, 17, 18] found that the wind influence at the top of the tower has a significant effect and tends partly to counterbalance the negative action at the inlet at high wind speeds. Furthermore the method proposed by Witte fails to give the correct velocity distribution through the heat exchanger for a horizontal arrangement [13].

Buxmann [12] and Völler [13] performed isothermal model tests to determine the effect of different outlet shapes of the cooling tower and the arrangement of the heat exchangers on the heat rejection rate of the tower in the presence of wind. To quantify the influence of wind on the air mass flow rate through the tower, Buxmann [12] defined pressure coefficients for the inlet, C_{pi}, and outlet, C_{po}, in terms of the static pressure difference between the throat of the tower and the ambient.

$$C_{pi}, C_{po} = \frac{\triangle p_w - \triangle p}{0.5 \rho v_w^2} \tag{1}$$

where $\triangle p_w$ is the static pressure difference with wind and $\triangle p$ is the static pressure difference without wind, for the same mean air mass flow rate through the tower. Their tests were all done for the same d_i/H_i ratio of the tower model and, except for the vertical arrangement of the heat exchangers, the pressure loss coefficient of the heat exchangers was kept at a constant value for all the tests. Furthermore Völler [13] determined the inlet pressure coefficient in a uniform cross-flow, but because a wind profile is usually found in the atmosphere, he assumed that the effective wind velocity which acts upon the tower inlet is equal to the average wind velocity across the inlet height of the tower. This assumption, however, was never confirmed by model tests. The velocity distribution through the heat exchangers in the presence of a cross-wind was measured separately and an air-side heat transfer coefficient correction factor was defined to calculate the reduction in heat transfer due to the flow distortions.

By employing the experimental results, Völler calculated the influence of the wind effect on two dry-cooling towers and compared the results with the measurements made on the full scale towers. Good agreement is found between the predicted values and the results as published by Van der Walt [3] for the Grootvlei 5 tower. For the Rugeley tower, with the heat exchangers arranged vertically around the circumference of the tower, the rise in water outlet temperature as calculated by Völler is considerably lower than that which was measured by Christopher [2].

Apparatus

Due to all the uncertainties, comprehensive scale model tests were performed in the present investigation. Specific reference is made to circular natural draft dry-cooling towers with finned tube heat exchangers arranged uniformly and horizontally (not A-frames) over the entire inlet section of the tower.

In the present investigation the effect of winds on the heat rejection rate of dry-cooling towers is studied by using isothermal model tests, similar to those used by Völler [13]. Two different models were used to simulate the shell of a dry-cooling tower. Each of these models had a sharp edged inlet of diameter 200 mm. For the first model, a cylindrical cooling tower was simulated by using a PVC pipe with an outside diameter of 200 mm and a wall thickness of 4 mm. For the second model, a 160 mm diameter pipe was used with the inlet shape of the tower simulated by a conical section attached to the one end of the pipe. The latter had a cone apex angle, $2\theta_c$, of 24° as shown in figure 6. The tower draft was simulated by connecting the other end of the tower model to the suction side of a radial fan. The air flow rate through the model was determined with the aid of an elliptical nozzle. All the tests were done with the axis of the model in a horizontal position and the ground level is therefore simulated by a vertical surface. By adjusting the distance between the ground level and the inlet edge of the model, the inlet pressure coefficient can be determined for different values of the d_i/H_i ratio of the tower. All the parts of the model can easily be removed or modified, thereby making it possible to determine the separate effect of each of the above-mentioned components on the inlet pressure coefficient.

Typical finned tube heat exchangers which are found in existing dry-cooling towers were simulated in the model with the aid of two pieces of honeycomb in series with additional resistances in the form of perforated plates or screens. The honeycomb is used to direct the air flow at the inlet of the model in an exial direction. The heat exchanger loss coefficient is defined as:

$$K_{he} = \frac{\triangle p_{he}}{0.5 \bar{\rho} v_{he}^2} \tag{2}$$

where $\triangle p_{he}$ is the difference in total pressure across the core, while v_{he} is the air velocity based on the frontal area of the heat exchangers. For the present arrangement of the heat exchangers v_{he} is also equal to the mean air velocity, v, based on the inlet cross-sectional area of the tower. In figure 1 the heat exchanger loss coefficient is shown with five different additional resistances in series with the honeycomb as a function of the characteristic flow parameter, Ry.

$$Ry = \rho \; v_{he} \; / \; \mu \tag{3}$$

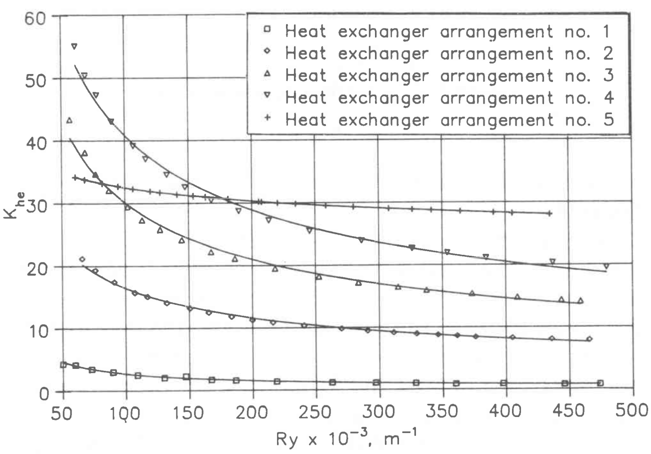

Figure 1: Heat exchanger loss coefficients

The cooling tower inlet loss coefficient, K_{ct}, in the absence of wind effects and tower supports, as defined by Du Preez and Kröger [19], was determined with the present model for different values of the d_i/H_i ratio of the tower. The results are shown in figure 2 as a function of the Reynolds number inside the model. The results of Du Preez and Kröger [19], obtained at a Reynolds number of 1.8×10^6, are also shown, suggesting that K_{ct} is independent of the Reynolds number for the range of the data shown in the figure 2. It was furthermore found that the inlet shape (taper) of the tower shell has no measurable effect on the value of the inlet loss coefficient, because the flow separates at the tower inlet edge and a free jet is formed as also observed by Russell [8].

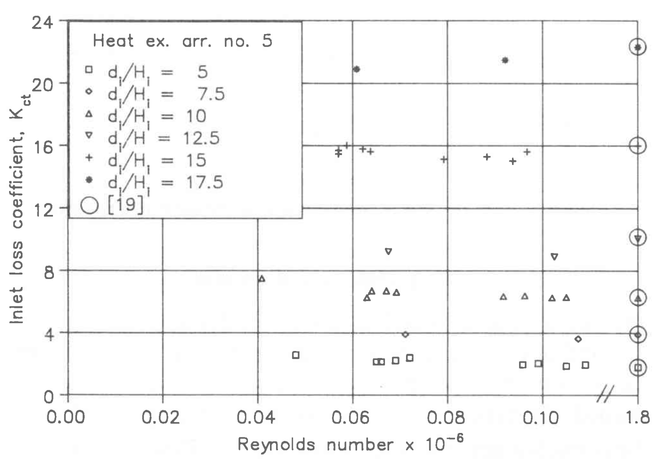

Figure 2: K_{ct} for a cooling tower with a horizontal heat exchanger arrangement

The supports on which the cooling tower shell and the horizontal heat exchangers rest should be modelled in such a way that the pressure loss coefficient of the latter is the same as that found in the full scale tower. An arbitrary, but representative, sets of supports was incorported in the tower model. The mean pressure loss coefficient of the supports, K_{ts}, can be expressed approximately in terms by the drag coefficient, C_{Dts}, i.e.

$$K_{ts} = C_{Dts} \; L_{ts} \; d_{ts} \; n_{ts}/(\pi \; d_i \; H_i) \tag{4}$$

where L_{ts} is the support length, d_{ts} is its effective diameter and n_{ts} is the number of supports. The drag coefficient for rectangular supports as commonly found in dry-cooling towers is essentially independent of the Reynolds number. Hoerner [20] lists the drag coefficients of such profiles, and for a square the latter is given as 2.05 in the range $10^4 \leq Re \leq 10^6$. Due to the influence of the supports on each other, the experimental value of K_{ts} tends to be higher than that found when the supports are considered as a row of single bodies in a free stream.

In the model the tower and heat exchanger supports were conveniently simulated by round metal rods having diameters of 1.5 and 3 mm respectively. Since the drag coefficient for a circular cylinder is a function of the Reynolds number, the approximate air velocity across the supports has to be known to obtain the corresponding drag coefficient. Völler [13] measured the velocity distribution in the vertical inlet section of a cooling tower for various values of the velocity ratio v_w/v. According to the measurements, the minimum and maximum Reynolds numbers across the supports in the present tests are 980 and 7580 respectively. In this range of Reynolds numbers, the drag coefficient for a cylinder has an average value of approximately 0.975. By rearranging equation (4), the number of supports needed in the model to obtain the same pressure loss coefficient as in the full scale tower can be calculated. The supports in the model are arranged uniformly in 5 concentric circles, with further details given in Table 1.

d/d_i	d_{ts} mm	K_{ts}	n_{ts}
1.000	3	0.362	76
0.918	1.5	0.0929	36
0.714	1.5	0.1195	36
0.497	1.5	0.1029	22
0.293	1.5	0.1906	22

Table 1: Tower supports in the model

Two different open wind tunnels were used to simulate the cross-flow. Because of the flexibility and availability of the smaller tunnel (330 mm × 660 mm), it was used for most of the tests. By using a second tunnel with a much larger outlet section (1710 mm diameter), shown schematically in figure 3, the influence of the blockage effect on the experimental results was determined. The blockage effect in an open jet will cause the effective velocity across the model to be less than that of the free stream. For both wind tunnels the effective wind velocity across the model was determined by using a solid blockage correction factor as determined by Lock [21]. The difference in the experimental results obtained on the two tunnels was found to be negligible.

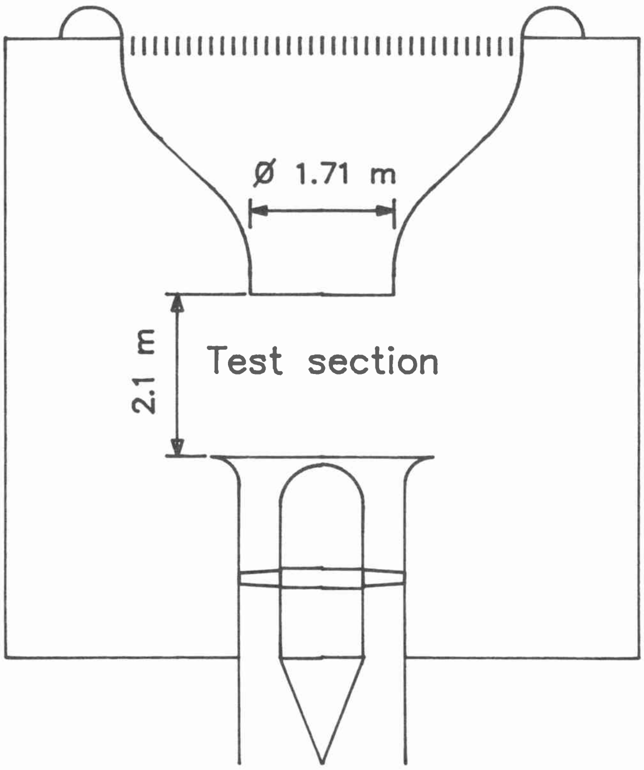

Figure 3: Layout of wind tunnel

To increase the effective Reynolds number based on the outside surface of the model, the latter was roughened with sandpaper. The height of the roughness elements, k, on the sandpaper was 0.35 mm and the relative roughness, k/d, was therefore 0.00175. For bluff bodies, such as circular cylinders, the value of the critical Reynolds number, for which the drag coefficient shows a sudden drop, depends among others on the roughness of the surface. Fage [22] performed model tests to determine the drag on roughened cylinders, and found that for a relative roughness of 0.00175, the critical Reynolds number is approximately 1.6×10^5.

During flow across a cylinder, the static pressure varies circumferentially. The pressure distribution around both the roughened and smooth cylinders was measured for different Reynolds numbers. A static pressure coefficient is defined as

$$C_p = 2(p_\theta - p_\infty)/\rho_\infty v_\infty^2 \tag{5}$$

where p_θ is the local static pressure and the other variables refer to the free stream. The minimum pressure coefficient, C_{pmin}, is a function of the wind Reynolds number

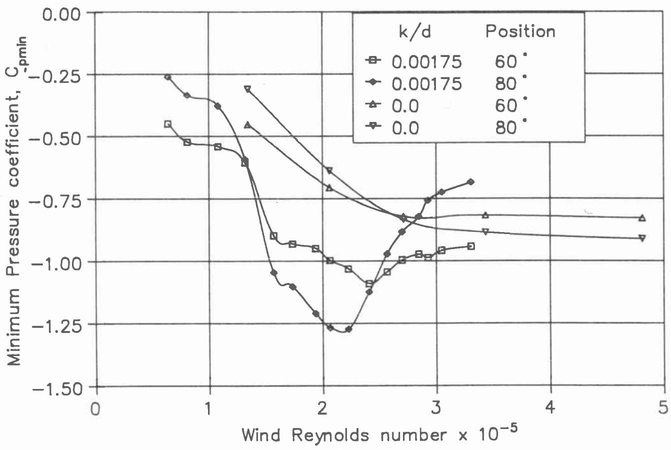

Figure 4: C_{pmin} for flow across a cylinder

and the circumferential position on the model as shown in figure 4.

In figure 5 the inlet pressure coefficient is shown as a function of the wind Reynolds number. The latter was determined with no flow through the model and the pressure difference between the surroundings and the inside of the tower is therefore only caused by the cross-wind. It is obvious from figures 4 and 5 that the static pressure inside the tower, and therefore also the inlet pressure coefficient, is a function of the wind Reynolds number. To eliminate this effect, all the tests in the present investigation were performed at a constant Reynolds number of roughly 4.8×10^5, except where stated differently. Since the critical Reynolds number for the tower model is 1.6×10^5, the present tests were performed in the supercritical range of Reynolds numbers.

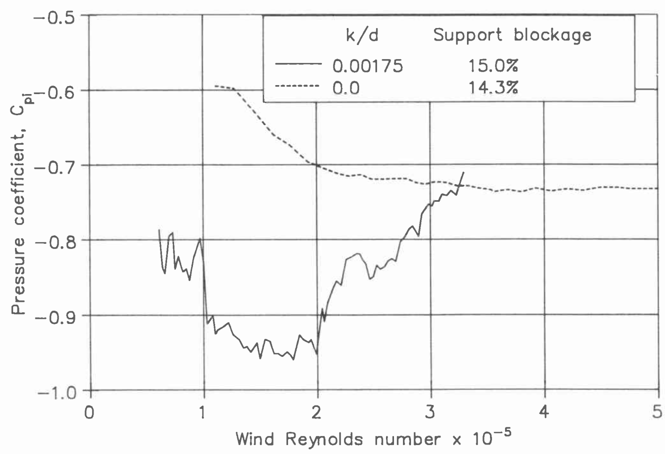

Figure 5: Inlet pressure coefficient with no flow in the tower

Experiment and results

By employing the cooling tower model described above, the influence of the different components of the cooling tower on the inlet pressure coefficient, C_{pi}, was determined. Specific reference is made to the d_i/H_i ratio, the heat exchanger and the tower support loss coefficients, the shape of the tower shell and the form of the approaching wind profile. It was found that the effect of the above-mentioned parameters on the value of C_{pi} are interdependent and it is therefore impossible to quantify the effect of these parameters individually.

Results of tests obtained with a uniform wind profile and tower supports are shown in figures 6, 7 and 8 as functions of the velocity ratio, v_{wo}/v, where v_{wo} is the wind velocity at the outlet height of the tower. Different values of v_{wo}/v were obtained by varying the air velocity inside the tower for a constant wind velocity.

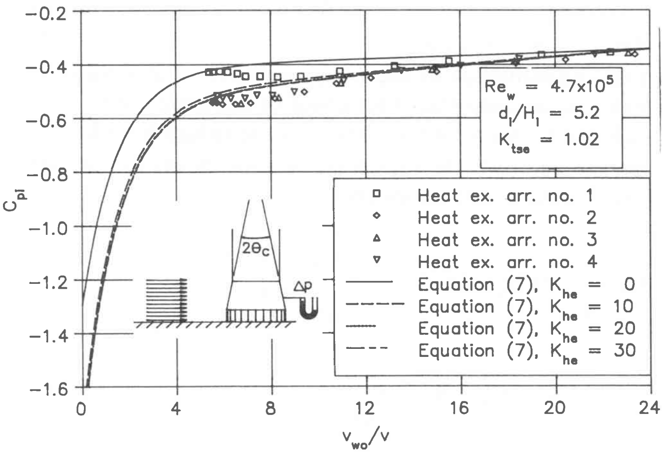

Figure 6: Inlet pressure coefficient for a d_i/H_i ratio of 5.2

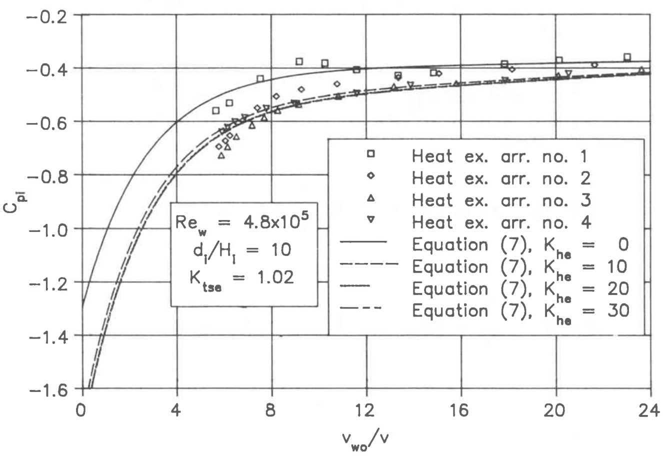

Figure 7: Inlet pressure coefficient for a d_i/H_i ratio of 10

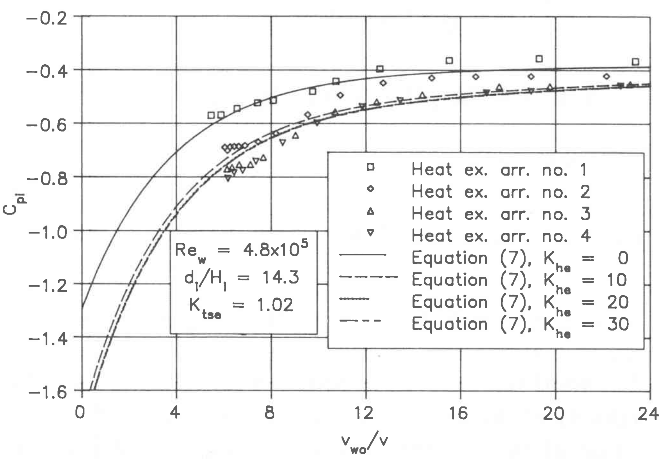

Figure 8: Inlet pressure coefficient for a d_i/H_i ratio of 14.3

The figures show that C_{pi} is almost independent of the d_i/H_i ratio for large values of v_{wo}/v. For smaller velocity ratios, the absolute value of C_{pi} increases slightly with an increase in the value of d_i/H_i. C_{pi} also increases for an increase in the heat exchanger pressure loss coefficient. It has also been found that the conical shape of the tower shell has no effect on C_{pi} if the present arrangement of the tower supports is installed at the tower inlet.

A test was also performed where the honeycomb, used to direct the flow in an axial direction, was removed and the horizontal heat exchangers were modelled by means of a number of mesh layers only. The inlet pressure coefficient obtained in the test was found to be the same as that in the previous test where the honeycomb was used in series with the mesh layers. It is therefore sufficient to model the horizontally aranged finned tube heat exchangers, found in dry-cooling towers (relatively high loss coefficient), only with mesh layers, if the inlet pressure coefficient in the presence of a cross-wind is to be determined.

In the atmosphere the wind profile is usually not uniform and significant variations in the wind velocity may be found near the ground surface. The wind profile can, for engineering applications, be described by a power law [23].

$$\frac{v_w}{v_{wref}} = \left(\frac{z}{z_{ref}}\right)^b \tag{6}$$

where v_{wref} is the wind velocity at the reference height z_{ref}. The exponent b depends above all on the morphology of the ground surface with the vertical temperature distributions also having an important effect. VDI 2094 [24] suggests that in the case of cooling towers, $b = 0.2$ and the reference height is chosen as the outlet height of the cooling tower. During the model tests 6 mm diameter dowels were arranged 450 mm upstream to create such a wind profile at the location of the model as shown in figure 9.

The results of model tests obtained with a wind profile are shown in figure 10 for different wind Reynolds numbers. The inlet pressure coefficient shown in the figure was determined by substituting the relatively high wind

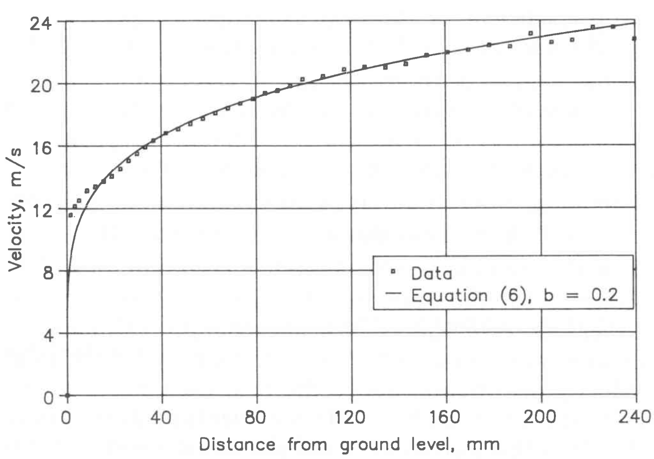

Figure 9: Wind profile

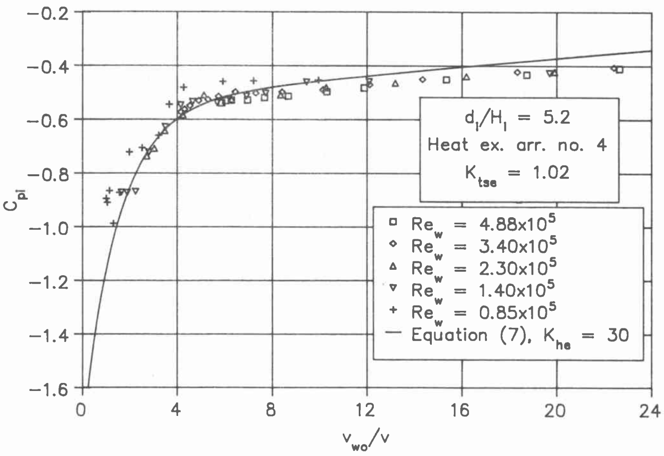

Figure 10: Inlet pressure coefficient

velocity at the outlet height of the cooling tower into equation (1). If the results obtained at the maximum Reynolds number are compared with those in figure 6, it would appear that the wind profile has almost no effect on the value of C_{pi}. It would seem that the inlet pressure coefficient is not only affected by the wind velocity over the tower inlet height, but also by the wind velocity at higher elevations. The assumption by Völler [13], that the average wind speed over the inlet height of the cooling tower should be used to calculate the wind effect on the inlet, is thus shown to be without foundation.

In order to determine the effect of the tower supports on C_{pi}, several tests were done without any supports installed in the tower inlet. Although the results obtained are only of academic interest, they can be used to determine the extreme value of C_{pi} as K_{ts} decreases towards zero. With the tower supports removed, the absolute value of C_{pi} increases considerably and also becomes a function of the wind profile and the taper angle of the tower shell. For example, with a d_i/H_i ratio of 5 and a uniform wind profile, C_{pi} increases roughly by 170% and 150% for a cooling tower with a taper angle, $2\theta_c$, of 0 and 24° respectively. For the wind profile shown in figure 9, C_{pi} increases by 120% and 70% respectively for the same tower geometry. These tests further indicate that for an increase in the d_i/H_i ratio, C_{pi} becomes more dependent on the pressure drop coefficient of the heat exchangers, especially for small values of v_{wo}/v. In this region, towers with large K_{he} values tend to be more sensitive to cross-winds.

A possible explanation for the latter can be found in data presented by Geldenhuys and Kröger [25], who determined K_{ct} for different d_i/H_i and K_{he} values. Their results suggest that for an increase in the value of K_{he} from zero to 30, K_{ct} remains almost constant for a d_i/H_i ratio of 5, while a significant reduction in K_{ct} is found for a d_i/H_i value of 15. Furthermore, the increase in the cooling tower loss coefficient as the heat exchanger resistance decreases, was found to be due to a tendency for the axial velocity distribution above the heat exchangers to become more non-uniform. Measurements made by Völler [13] show that a cross-wind will also cause a non-uniform axial velocity distribution in a tower. According to the definition of C_{pi} in equation (1), the latter gives an indication of the increase in the value of the inlet loss coefficient, K_{ct}, for a cooling tower in a cross-wind. It would therefore be expected that a cooling tower with a poor velocity distribution in windless conditions will be less affected by cross-winds than a tower with a more uniform distribution and a consequent small inlet loss coefficient in windless conditions. The results of the model tests seem to confirm this.

Due to the interrelation between the different parameters on the value of C_{pi} as discussed above, it is impossible to find a simple correlation for C_{pi}. If it is accepted that the influence of the tower supports, the form of the wind profile and the taper angle of the tower shell on the value of C_{pi} vary linearly between the values for which the tests were done, the following relation holds for cylindrical shaped tower supports.

$$
\begin{aligned}
C_{pi} = \Bigg[& -0.57 + 0.0503\left(\frac{v_{wo}}{v}\right)^{0.8}\left(\frac{d_i}{H_i}\right)^{-0.64} \\
& -1.2/exp\left(2.4\left(\frac{v_{wo}}{v}\right)\left(\frac{d_i}{H_i}\right)^{-0.8}\right)\Bigg] \\
& \times\left[1-\left\{\frac{0.0067}{exp\ (0.2\ K_{he})}\right\}\left\{40-6\left(\frac{v_{wo}}{v}\right)\left(\frac{d_i}{H_i}\right)^{-0.8}\right\}\right] \\
& +\Bigg[\Bigg[-0.6+0.01\left(\frac{d_i}{H_i}\right)+\Bigg\{\text{-}0.65+0.06\left(\frac{d_i}{H_i}\right) \\
& +0.1K_{he}\times\Bigg(0.23-0.039\left(\frac{d_i}{H_i}\right) \\
& +0.001\left(\frac{d_i}{H_i}\right)^2\Bigg)\Bigg\}\times 0.054\left(24-\frac{v_{wo}}{v}\right)\Bigg] \\
& +sin\left\{\frac{v_{wo}}{v}\ /\left(1+\frac{0.17v_{wo}}{v}\right)\right\}\ /exp\left\{\frac{v_{wo}}{7v}\right. \\
& \left.+0.2\left(15-\frac{d_i}{H_i}\right)+\frac{K_{he}}{20}\right\}\Bigg] \\
& \times(1-0.978K_{tse})\{1-(0.003\times 2\theta_c+2b \\
& +0.027\times 2\theta_c\times b)\}
\end{aligned} \tag{7}
$$

$$
for\ 5 \le \frac{d_i}{H_i} \le 15\ ,\ 0 \le K_{he} \le 30\ ,\ 0 \le \frac{v_{wo}}{v} \le 24
$$

$$
0 \le b \le 0.2\ ,\ 0 \le 2\theta_c \le 24°\ ,\ 0 \le K_{tse} \le 1.02
$$

The wind velocity, v_{wo}, in equation (7) refers to the wind velocity at the top of the cooling tower, while K_{he} is the value of the pressure loss coefficient of the heat exchangers obtained at an arbitrary Ry value of 200×10^3.

The effective pressure loss coefficient of the tower supports, K_{tse}, is defined as the sum of the K_{ts} values of the different rings of supports based on the circumferential inlet area of the cooling tower. This equation is compared to the experimental results in figures 6, 7, 8 and 10.

As the value of K_{tse} approaches the upper limit of 1.02, the second term in equation (7) reduces to zero and C_{pi} becomes independent of the wind profile and the contraction angle of the cooling tower shell. The results in figures 6, 7 and 8 furthermore suggest that for high values of K_{tse}, C_{pi} also becomes independent of the values of K_{he} for sufficiently high values. Therefore equation (7) is also applicable to cooling towers with K_{he} values higher than 30, provided that K_{tse} approaches 1.02.

A cross-wind distorts the velocity distribution through the heat exchangers [13]. In the present investigation the velocity distribution was determined only for a horizontal arrangement of the heat exchangers, with the cylindrical tower model being used in all the tests. For this purpose four total pressure tubes were positioned 20 mm above the heat exchanger in such a way that the representative cross-sectional area was the same for each pressure tube. Due to symmetry the velocity distribution was only measured in one half of the model.

Figures 11, 12 and 13 show the ratio of the local air velocity, v_l, to the mean air velocity through the heat exchangers for different values of the relative wind velocity v_{wo}/v. As the wind speed increases, the flow through the heat exchanger becomes increasingly more non-uniform with the maximum air velocity at the lee side of the tower. On the upstream side the air flow through the heat exchanger decreases, due to the formation of a separated flow pattern below the heat exchangers. In contrast to the inlet pressure coefficient, the cylindrical tower supports have no effect on the velocity distribution. Further tests revealed that for towers with small values of the heat exchanger pressure loss coefficient, such as commonly found in wet towers, the velocity distribution becomes more non-uniform.

The distorted air flow pattern through the finned tube heat exchangers will influence the heat transfer characteristics of the heat exchanger. For a uniform air velocity distribution the heat transfer rate in a cross-flow finned tube heat exchanger with two or more tube passes (essentially counterflow), commonly found in dry-cooling towers, can be approximated by

$$Q \approx v\rho_a A_{fr} c_{pa}(T_{wi}-T_{ai})\epsilon \tag{8}$$

with the effectiveness, ϵ, given by Holman [26] for a counterflow geometry. If the air velocity through the heat exchangers is distorted, the reduced effective heat transfer rate is found by introducing a correction factor such that

$$Q_e = \alpha_Q Q$$

where

$$\alpha_Q = \int_{A_{fr}} v_l \rho_a c_{pa} \epsilon \, dA_{fr} / (v\rho_a A_{fr} c_{pa}\epsilon) \tag{9}$$

The heat transfer correction factor is a function of the overall heat transfer coefficient of the heat exchangers which may be approximated by

$$UA = a_{UA} Ry^{b_{UA}} \tag{10}$$

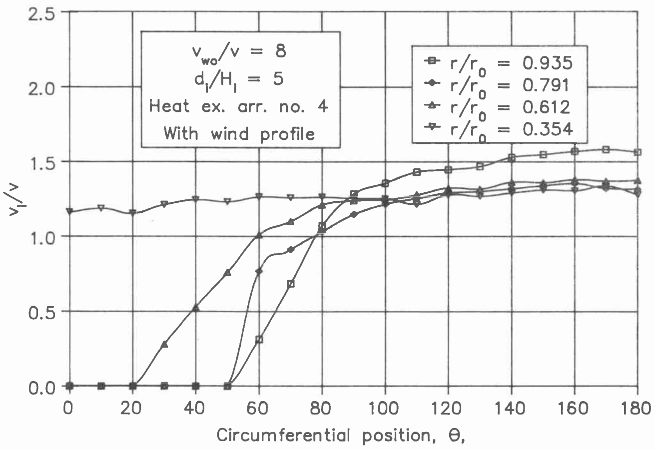

Figure 12: Velocity distribution through the heat exchanger, $v_{wo}/v = 8$

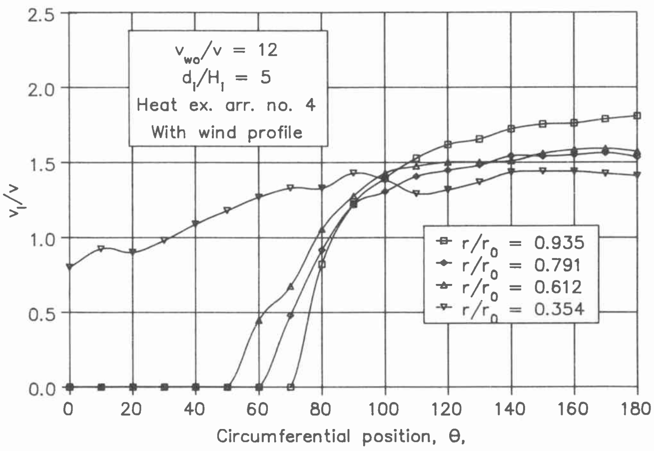

Figure 13: Velocity distribution through the heat exchanger, $v_{wo}/v = 12$

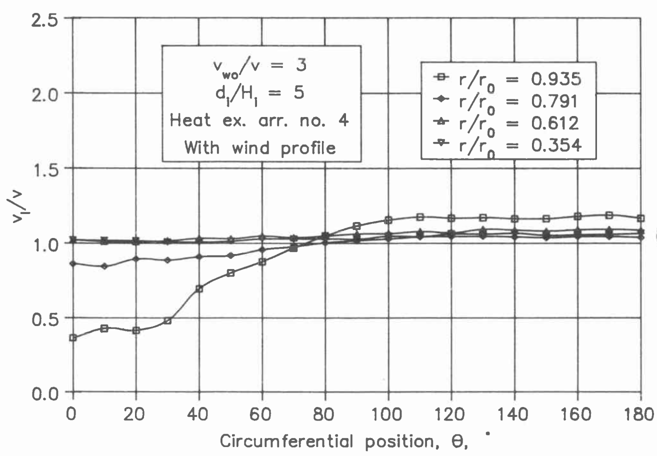

Figure 11: Velocity distribution through the heat exchanger, $v_{wo}/v = 3$

As the value of b_{UA} increases an increase in the value of α_Q is found. The exponent b_{UA} is a characteristic of the type of heat exchanger that is used, however, for the finned tube heat exchangers commonly found in dry-cooling towers the value of b_{UA} is in the order of 0.45. Based on the measured velocity distribution through the horizontal heat exchanger, α_Q was calculated for different relative wind velocities and is shown in figure 14. The following empirical relation is recommended for design purposes within the limits specified below and is compared to the experimental results in figure 14.

$$\alpha_Q = \left\{ 1 - \frac{\left(\frac{v_{wo}}{v}\right)^{3.6} exp\left(-\frac{v_{wo}}{3.75v}\right) v^{0.576}}{3561} \right\}$$

$$\times \left[0.98 + 0.02 \left\{ exp\left(5.2 - \frac{d_i}{H_i}\right) + exp\left(-\frac{v_{wo}}{v}\right) \right\} \right]$$

$$-(1.5 - 0.05K_{he}) \times \quad \left[\left\{ 0.013 - 0.0048\left(\frac{d_i}{H_i}\right) \right. \right.$$

$$\left. + 0.000302\left(\frac{d_i}{H_i}\right)^2 \right\}$$

$$+ \left\{ 0.0134 \; - \; 0.00129\left(\frac{d_i}{H_i}\right) + 0.000038\left(\frac{d_i}{H_i}\right)^2 \right\} \times \left(\frac{v_{wo}}{v}\right)$$

$$\left. + \left\{ 0.0035 + 0.00206\left(\frac{d_i}{H_i}\right) - 0.000085\left(\frac{d_i}{H_i}\right)^2 \right\} sin\left(\frac{v_{wo}}{1.9v}\right) \right]$$

$$-\left(0.0053 - \frac{K_{he}}{9182}\right)(11.26 - 25.64 b_{UA})\frac{v_{wo}}{v} \qquad (11)$$

$$for\ 5.2 \leq \frac{d_i}{H_i} \leq 15 \ , \ 10 \leq K_{he} \leq 30 \ , \ 0 \leq \frac{v_{wo}}{v} \leq 12$$

$$0 \leq b \leq 0.2 \ , \ 0 \leq K_{tse} \leq 1.02$$

$$0.4 \leq b_{UA} \leq 0.5 \ , \ 1 \leq v \leq 4 \ m/s$$

Note that if $\alpha_Q > 1$ in the above equation a value of $\alpha_Q = 1$ is assumed.

The distorted air flow pattern through the heat exchangers will furthermore cause a non-uniform air temperature distribution in the tower which will affect the available tower draft. A further problem which arises in this regard, is that the amount of mixing inside the tower due to large scale vortices is not known; consequently an assumption has to be made. One extreme would be to assume that the air just above the heat exchangers is perfectly mixed. A mass mean air temperature would therefore be used to calculate the tower draft and a somewhat optimistic result will be obtained. The other extreme

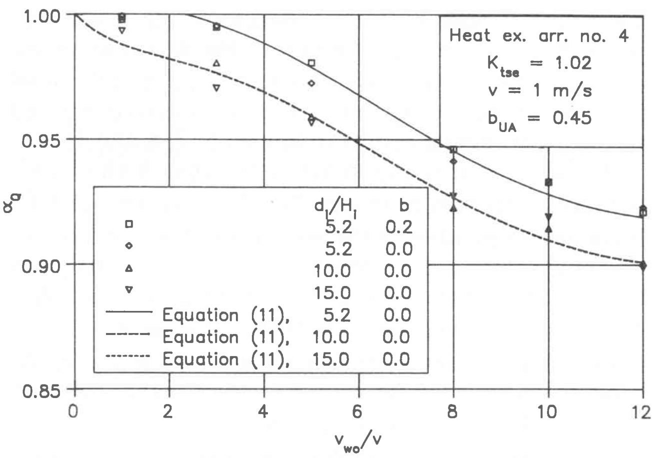

Figure 14: Heat transfer correction factor

would be to allow no mixing at all, and in this case the coldest air temperature inside the tower has to be employed to calculate the draft, resulting in a pessimistic solution. The minimum temperature of the air leaving a heat exchanger which is subjected to a non-uniform velocity distribution is associated with the maximum air velocity through it. Based on experimental measurements, equation (12) is proposed to obtain the ratio of the maximum air velocity through a horizontally arranged heat exchanger to the mean velocity through it. The minimum air outlet temperature, and therefore also the available tower draft, can be calculated for known water and air inlet temperatures and heat exchanger characteristics.

$$\frac{v_{max}}{v} = 1.014 - 0.0095\left(\frac{d_i}{H_i}\right) + 0.0014\left(\frac{d_i}{H_i}\right)^2$$

$$+ \left\{ -0.1265 + 0.0509\left(\frac{d_i}{H_i}\right) - 0.00245\left(\frac{d_i}{H_i}\right)^2 \right\} \frac{v_{wo}}{v}$$

$$\times (1 - 0.26b)$$

$$+ \left[-0.362 + .0865\left(\frac{d_i}{H_i}\right) - 0.00321\left(\frac{d_i}{H_i}\right)^2 \right.$$

$$\left. + \left\{ 0.288 - 0.0572\left(\frac{d_i}{H_i}\right) + 0.00242\left(\frac{d_i}{H_i}\right)^2 \right\} \frac{v_{wo}}{v} \right]$$

$$\times (1.5 - 0.05K_{he}) \qquad (12)$$

$$for\ 5.2 \leq \frac{d_i}{H_i} \leq 15 \ , \ 10 \leq K_{he} \leq 30 \ , \ 0 \leq \frac{v_{wo}}{v} \leq 12$$

$$0 \leq b \leq 0.2 \ , \ 0 \leq K_{tse} \leq 1.02$$

The wind effect at the outlet of the cooling tower has been studied by a number of investigators [10, 11, 12, 13, 15, 16, 17, 18]. In the present study the wind effect on only a cylindrical cooling tower outlet was studied, using the same tower model. A few mesh layers were placed one diameter from the outlet edge of the model to create a uniform velocity distribution in that position. The tests were repeated for different velocity ratios by varying the air velocity inside the model for a constant wind speed, with the results shown in figure 15. The data obtained at the maximum Reynolds number for a cylindrical tower outlet, i.e. with the outlet to throat area ratio A_o/A_t equal to 1, are correlated by equation (13), and are also shown in figure 15 in its range of applicability. By using data for divergent and convergent outlet shapes as published by Völler [13], the equation obtained for a cylindrical cooling tower outlet was extended to other shapes as well within the limits as specified.

$$C_{po} = -0.405 + 1.07\left(\frac{v_{wo}}{v}\right)^{-1}\left(\frac{A_o}{A_t}\right)^{-1.65}$$

$$+ 1.8\, log_{10}\left\{\left(\frac{v_{wo}}{2.7v}\right)\left(\frac{A_o}{A_t}\right)^{1.65}\right\}$$

$$\times\left\{\left(\frac{v_{wo}}{v}\right)\left(\frac{A_o}{A_t}\right)^{1.65}\right\}^{-2} + \left\{-1.04 + 1.702\left(\frac{A_o}{A_t}\right)\right.$$

$$\left. -0.662\left(\frac{A_o}{A_t}\right)^2\right\} \times \left(\frac{v_{wo}}{v}\right)^{-0.7} \quad (13)$$

$$for\ 1.8 \le \frac{v_{wo}}{v} \le 24\ if\ \frac{A_o}{A_t} = 1\ ;\ 1.8 \le \frac{v_{wo}}{v} \le 12\ if\ \frac{A_o}{A_t} \ne 1$$

$$0.893 \le \frac{A_o}{A_t} \le 1.232$$

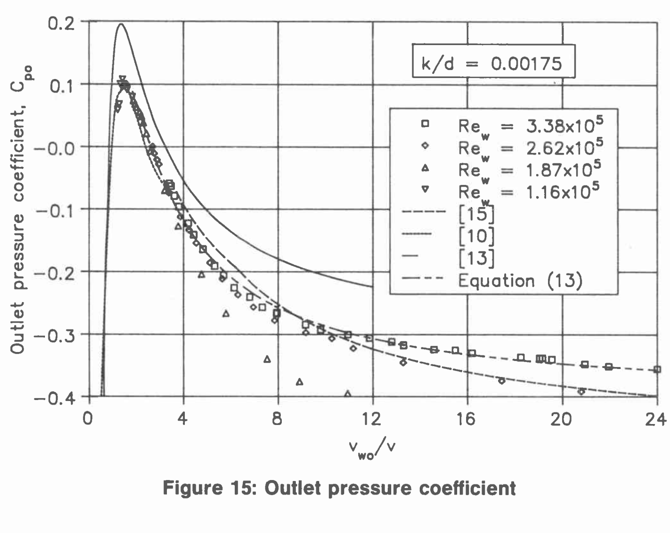

Figure 15: Outlet pressure coefficient

With isothermal model tests, the effect of the wind on the inlet and outlet of the tower is normally tested separately. It is assumed that the wind effect on the tower outlet is independent of that at the inlet and vice versa. Figures 11-13 show that the wind effect at the inlet causes a non-uniform velocity distribution in the tower. The influence of this flow pattern on the outlet pressure coefficient was also investigated. This velocity distribution was modelled with the aid of a flow resistance with a non-uniform loss coefficient placed one diameter from the outlet edge inside the model. It was found, however, that this velocity distribution had no effect on the outlet pressure coefficient. Furthermore the wind effect on the tower outlet also had almost no influence on the flow pattern inside the tower.

By using the experimental results shown in the previous figures, the rise in the outlet water temperature of a cooling tower with a fixed heat rejection rate can be calculated for different wind speeds. For this purpose, equations (7), (11), (12) and (13) were employed in a computer program to predict the behaviour of a specific cooling tower in a cross-wind. The computer program was developed to determine the operating point of a given cooling tower with a horizontal heat exchanger layout. This is done by following an iterative procedure to obtain the value of the air mass flow rate through the tower that will satisfy both the energy and draft equations. With a known air mass flow rate and rise in air temperature, the heat rejection rate of the tower can be calculated.

If, however, the heat rejection rate of the tower is kept constant for different atmospheric conditions, the corresponding water inlet temperature can also be obtained with the program. This is done by introducing another iterative procedure, using an interval halving search method for finding the correct water inlet temperature.

Figure 16 shows the effect of a cross-wind on an example of a natural draft hyperbolic, concrete cooling tower. The outlet height of the tower is 120 m while the inlet height and inlet diameter are 13.67 m and 78.3233 m respectively. Furthermore are the pressure drop and heat transfer characteristics of the heat exchangers employed in the tower respectively given by:

$$K_{he} = 1383.94795\ Ry^{-0.332458}$$

$$Ny = 383.61731\ Ry^{0.523761}$$

where Ny is the characteristic heat transfer parameter as defined by Kröger [27]. It is furthermore assumed that the wind profile can be described by the power law, with the exponent b in equation (6) equal to 0.2. The heat rejection rate of the cooling tower is kept at a constant value of 354.39 MW for all wind speeds, while the water mass flow rate through the tower = 4390 kg/s.

The curves in figure 16 can be compared with similar results as presented by Buxmann [14]. In the first curve shown in the figure, only the suction effect of the wind on the tower inlet was considered. As the wind speed increases, the wind effect on the inlet causes the available tower draft to reduce. As a result, the approach temperature of the tower increases to maintain the same heat rejection rate of the tower.

In the second curve, the wind effect on the tower outlet is also included. For wind speeds higher than 12 m/s, the wind influence on the outlet has a favourable effect on the tower performance, thus reducing the increase in the ap-

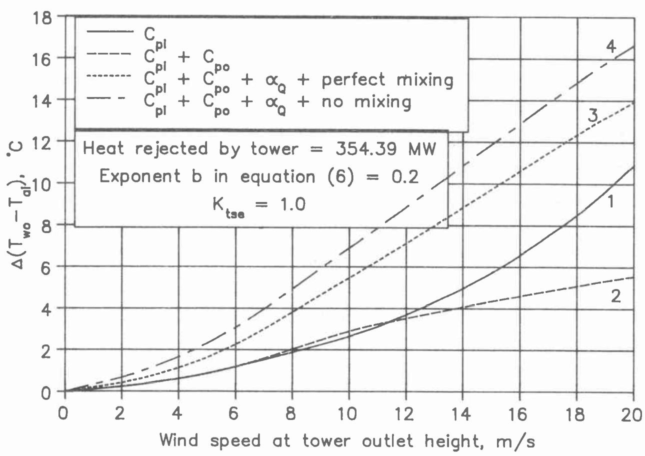

Figure 16: Rise in the approach water temperature as a function of the wind speed

proach temperature of the tower. Conversely for wind speeds between 7 and 12 m/s, the tower performance deteriorates.

In the third and fourth curve, the non-uniform velocity profile through the heat exchangers is also included. This causes a reduction in the effectiveness of the heat exchanger, as well as a non-uniform temperature distribution in the tower. In the third curve the air is assumed to the perfectly mixed, while in the fourth curve no mixing is assumed. The correct answer is expected to be somewhere between the two above-mentioned extremes, depending on the effectiveness of the mixing process.

Significant changes in the wind effect on the tower are found for a variation in the tower height. Due to the additional draft, higher towers tend to be less affected by cross-winds than those with a smaller height. Figure 17 shows the change in the approach temperature for a constant wind speed and water mass flow rate as a function of the tower height. The heat rejection rate of the tower was kept at a constant value of 354.39 MW by varying the initial water inlet temperature for each value of the tower height.

Further calculations revealed that for the heat exchangers arranged horizontally in the inlet cross-section of the tower the wind influence on the tower increases for an increase in the heat rejection rate of the tower.

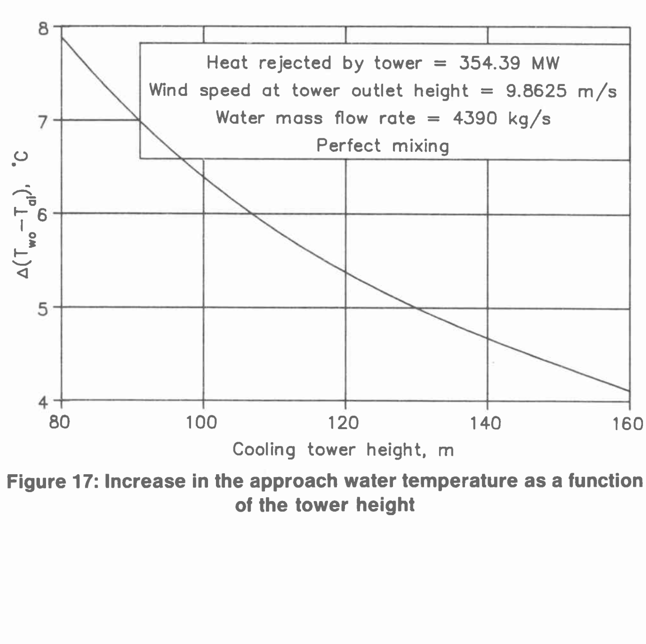

Figure 17: Increase in the approach water temperature as a function of the tower height

Conclusions

The reduction in the performance characteristics observed in full scale natural draft dry-cooling towers in the presence of a cross-wind was investigated by employing isothermal scale model tests. The inlet pressure coefficient is found to be strongly dependent on the pressure loss coefficient of the tower supports, the pressure loss coefficient of the heat exchangers and the inlet diameter to inlet height ratio of the tower. Generally it would appear that a tower becomes less sensitive to the wind for a reduction in the K_{he} value of the heat exchangers and a reduction in the d_i/H_i ratio of the tower. It is furthermore found that the taper angle of the tower shell and the form of the approaching wind profile have a very small effect provided that the pressure loss coefficient of the tower supports is sufficiently large. The wind effect on the tower also becomes increasingly less for an increase in the tower height. The tower draft is enhanced at higher wind speeds due to increased suction at the outlet of the tower.

Further reductions in the heat rejection rate of the tower in a cross-wind are caused by the distorted flow pattern through the heat exchangers. This is due to the reduction in the effectiveness of the heat exchanger and a non-uniform air temperature distribution in the tower. The air velocity distribution through the heat exchangers becomes increasingly more non-uniform as the d_i/H_i ratio of the tower increases and also for a reduction in the pressure loss coefficient of the heat exchangers.

Acknowledgement

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